Towards efficient system-level simulation and design of modern mechanical transmissions

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Abstract
Latest trends in the industry of mechanical transmissions demands for a new set of design tools, able to work beyond the assumption of the traditional analytical formulation. Recent progress in the field of model order reduction schemes finally made feasible to deal with dynamic 3D simulations of contact problems for both gears and bearing components, thus posing the base for a new frontier of simulation capabilities where different operational conditions can be analyzed. This paper aims at integrating some of these techniques into a commercial multi-domain 1D simulation software package, moving the first steps towards efficient system level analysis of typical 3D effects like micro/macro geometry modification, misaligned mounting, and perturbed loading conditions. A first proof-of-concept implementation is discussed along the paper. Furthermore, a broad set of case studies is presented, in order to showcase the main advantages of the proposed methodology, with respect to available commercial solution.

1 Introduction

Recent trends in mechanical transmission design is dominated by a continuous seek of weight reduction, with the joint and conflicting ambition to improve efficiency and reliability, as well as reducing the time-to-market. In such extreme scenario, assumptions behind traditional analytical formulations [1, 2] become questionable.

As a result, an increasing interest is being addressed towards the pursuit of improving current modelling strategies of the main transmission components (gears, shafts, bearings, and housing) in order to understand and effectively capture all potential sources of deviation from nominal operational conditions: flexibility, micro/macro geometry modifications of gears profiles, misaligned mounting, and perturbed loading conditions. To this aim, some recent research [3] demonstrated the potential of model order reduction (MOR) schemes in solving contact problems efficiently still allowing to preserve accuracy on derived quantities such as stress fields. Similar MOR schemes were exploited later for use in dynamic 3D simulations of contact problems for both gears [4] and bearing components [5].

Beside the above mentioned increased level of complexity, the possibility to assess early upfront sensitivity and/or optimization studies of the increased set of design parameters is crucial to improve the overall design methodology effectiveness. To this end, this paper deals with the ambition of integrating some of the above mentioned techniques into a commercial multi-domain simulation software package (SIEMENS LMS Imagine.Lab Amesim), thus widening the design toolset regarding transmission components.

Guided by the above stated goal, this papers proposes to exploit the quasi-static assumption in order to sacrifice accuracy in favor of a reduced computational complexity. As a result, the analysis of micro/macro geometry modifications, as well as shafts and gear body flexibility, under different/variable loading conditions, and possibly not ideal mounting, and their effects on other components of the transmission can be afforded within reasonable time. This is achieved by storing different simulations scenarios produced in a preprocessing phase by means of high order models, and then effectively interpolating among them to obtain a good estimation of the run-time operating condition.

The remainder of the paper is organized as follow: Section 2 describes the methodological background as well as the mathematical foundation of the proposed approach, while in section 3 a collection of four different application is discussed in order to emphasize the versatility as well as the main limitations of the approach. Section 4 concludes the paper

2 A system level 3D-1D integration framework

As mentioned in the introduction, the goal is to embed detailed information originating from high-fidelity nonlinear Finite Element (FE) simulations of gear pairs in a multi-domain environment for system level analysis of mechatronic drivelines. The overall process is realized in two different stages. The pre-processing stage will generate a set of Look-Up-Tables (LUTs), each capturing a different gear pair, under a set of desired working conditions (i.e. different torque levels and different angular configurations). During the processing stage, a model of transmission is assembled in LMS Imagine.Lab Amesim by using a combination of the standard submodules (driving signals generator, shaft flexibilities, roational compliances, stiffness, etc…), as well as the submodules specifically developed in this work, whose role is to interpolates between the values of the LUTs to generate the contact force corresponding to a given angular pitch. Section 2.1 and 2.2 will give more insight about the realization of the two above mentioned stages.

2.1 Pre-processing modules

This sections deals with the crucial aspects related to the FE framework used along this papers in order to numerically produce the *dictionary* of LUTs required to capture the behavior a specific gear pair under the desired operational conditions. In particular, the algorithm for the generation of one LUT consists in the sequential execution of the following three steps: (i) generation of the FE mesh; (ii) FE Static analysis for every gear pair; and (iii) computation of the LUT. Details for each of the steps are given in the corresponding subsections.

2.1.1 Generation of the FE mesh

In order to study effectively the effects of micro/macro geometry modifications, adequate attention need to be provided during the definition of the FE mesh. ISO standards [6, 7] provide the specifications of all the design parameters required to identify univocally the geometry of a specific gear and its manufacturing process.

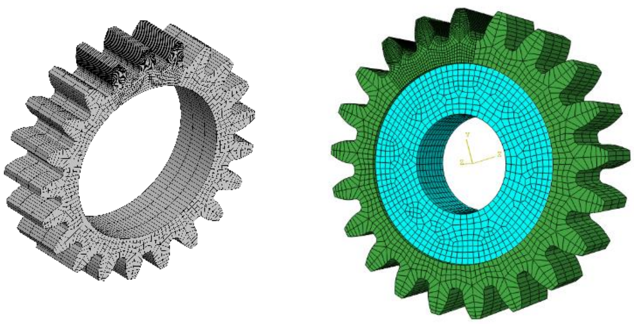


Figure 1: Generation of the FE mesh for non-standard gear body

For the purposes of this work, we relied on the in-house code MUTANT – Multibody Transient Analysis of Transmissions, available at the KUL PMA department [8]. In particular, two different strategies were developed depending on whether a non-standard gear body was required. In facts, for gears with a regular body using the mesher provided by MUTANT [8] sufficed the scope, while for more complex gears bodies (as the one illustred in Figure 1-right), usage of a mix of tools has been required. In such more complex cases, the mesher of MUTANT has been still used for the generation of the gear crown of teeth (in Figure. 1-left), whereas the non-standard body (colored in light blue in Figure 1-right) has been meshed separately, by using the auto-mesher functionality of the Abaqus commercial FE package. Thereafter, the two individual mesh sets are joined by eliminating the redundant dofs constraints based on the components shape functions.

2.1.2 MOR techniques for static analysis of gear contact

The contact mechanics problem is solved statically in order to derive a kinematic relation between the torque and the angular position of the gears.

$$\begin{bmatrix} 0 & 0 & 0 \\ & K_{FE}^1 & 0 \\ sym. & & K_{FE}^2 \end{bmatrix} \begin{Bmatrix} \theta_1 \\ u_f^1 \\ u_f^2 \end{Bmatrix} = \begin{Bmatrix} T^1 \\ 0 \\ 0 \end{Bmatrix} + Q_C \tag{1}$$

As already proposed in [4], the solution of system of equation in (1), corresponds to the equilibrium condition of two gears in contact under a given external load and for a given angular position. In particular, T represent the applied torque, Q_C the contact force vectors and K_{FE}^1 the stiffness matrix of the i^{th} gear. The unknown of the system correspond to θ_1 and u_f^i, which are respectively the angular position of the driving gear and the deformation pattern of each gear.

A feasible strategy to solve such a system consists in fixing the driven gear (i.e. gear 2) while the external torque is applied to the driving gear (i.e. gear 1): the static equilibrium condition is met when the elastic deformation of the gears balance the effects of the external torque, but also the contact forces. In particular, it is worth noting that at the equilibrium the “contact” torque, calculated by projecting the contact forces on the gear rotational degree of freedom, is equal to the external applied torque.

After solving (1), it is possible to evaluate the Static Trasmission Error (STE), $\Delta\theta$, defined as the difference between the angle of the driving gear in cinematic condition and the current angle of the driving gear, owing to the gear deformation under the applied load.

For calculating the contact forces, a penalty formulation has been adopted, where the penalty factor is chosen accordingly to the instruction given by [9], hence at least two orders of magnitude higher than the gear stiffness dimensions.

In order to speed-up the computation, a Model Order Reduction (MOR) scheme is used to reduce the size of the contact problems. More conspicuously, a reduction space Φ is assembled based on the possible nodes (and the corresponding DOFs) that can go into contact during the gear meshing process. Such space contains only one attachment mode for each DOF that can be possibly loaded during the meshing process and it is exploited such that:

$$q_i = \Phi\, u_f^i \tag{2}$$

$$K_i = \Phi'\, K_{FE}^i \Phi \tag{3}$$

By subsituting (2) and (3) into (1), we get

$$\begin{bmatrix} 0 & 0 & 0 \\ & K_i^1 & 0 \\ sym. & & K_i^2 \end{bmatrix} \begin{Bmatrix} \theta_1 \\ q_1 \\ q_2 \end{Bmatrix} = \begin{Bmatrix} T^1 \\ 0 \\ 0 \end{Bmatrix} + \widetilde{Q_C} \tag{4}$$

Where $\widetilde{Q_C}$ represents the projected contact forces on the reduction space:

$$\widetilde{Q_C} = \Phi\, Q_C \tag{5}$$

With the above described MOR scheme, the computational burden is drastically reduced, due to change of the linear system dimensionality decreased from about 10^(6), corresponding to the gear FE model DOFs number, to few thousands [4].

2.1.3 Filling of the LUT dictionary

As already mentioned at the beginning of Section 2.1, a dictionary of LUTs is required, with each LUT capturing a different set of working condition, for a particular gear pair. Therefore, the methodology discussed in the previous section is applied for retrieving the static equilibrium (4) of the desired gears for various angular positions and for different external loads.

In particular, exploiting the periodic symmetry of the gear contact interactions, it is sufficient to limit the range of the angular position to one angular pitch, which is divided in a fixed number of position, N_p. Moreover, a limited number of driving Torque values is used, N_t. Finally, the LUT is filled by storing the STE according to the following:

$$\Delta\emptyset(it, ip) = \theta_1 - \theta_2 \frac{r_2}{r_1}, \quad it = [1 \ldots N_t] \quad i_p = [1 \ldots N_p] \tag{5}$$

Where θ_1 and θ_2 are the angular position of the driving and the driven gear respectively; whereas r_1 and r_2 are the radii of the pitch circles. As a result, the LUT is stored as a matrix of dimensions $N_t \times N_p$

2.2 Processing modules

This section focus on the integration of the proposed methodology into a commercial simulation software package (SIEMENS LMS Imagine.Lab Amesim). Amesim is a bond-graph based multi-domain modeling software meaning that explicit data flow is promoted by assigning a causal connection between inputs and outputs. The inputs and outputs are grouped into ports which need to be compatible with the existing sub-models in the library. The benefit of such an approach is the modularity allowing the gear contact element to be reused several times within a single system level model and that a different description can be adopted without changing the overall system model. As the bond-graph method is energy based the port always consist of an effort and flow variable where the product represents the instantaneous power flowing through the system. As power is a scalar quantity, sub-models of different domains can be coupled.

In particular, a gear pair in a bond-graph environment is generally represented by a gear stiffness submodule with the equivalent inertia of the respective gears on both sides. For the purposes of this paper, two submodules have been developed: (i) the gear contact submodule, and (ii) the related gear inertia submodule. Both submodules have been modeled in a lumped fashion, i.e. neglecting any dynamics.

The first step to take for the generation of an Amesim block is to define it as an equivalent bond-graph component. The different components are: effort and flow sources, inertia's, capacitors, dissipators, junctions, transformers and gyrators. For a torque-displacement relationship with no dynamics, a capacitive element (i.e. spring) is deemed the best choice. The respective gear inertia's are uncoupled from the gear contact element and represented in the nearby components.

This is illustrated by a simple single gear stage model as shown in Figure 2. The first gear at the bottom is driven by an ideal torque source. The power is transmitted to the second gear via the contact force element. In the end of the chain, a damper which is fixed to the ground is connect to the second gear.

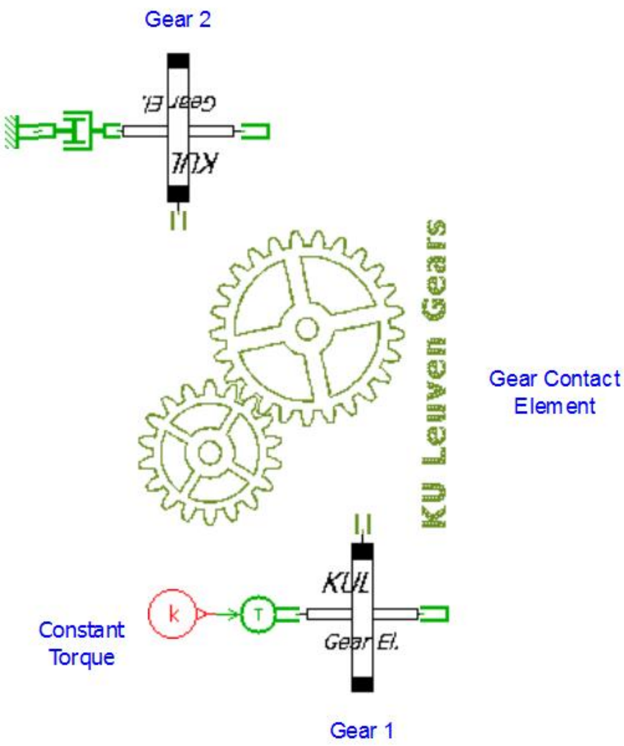
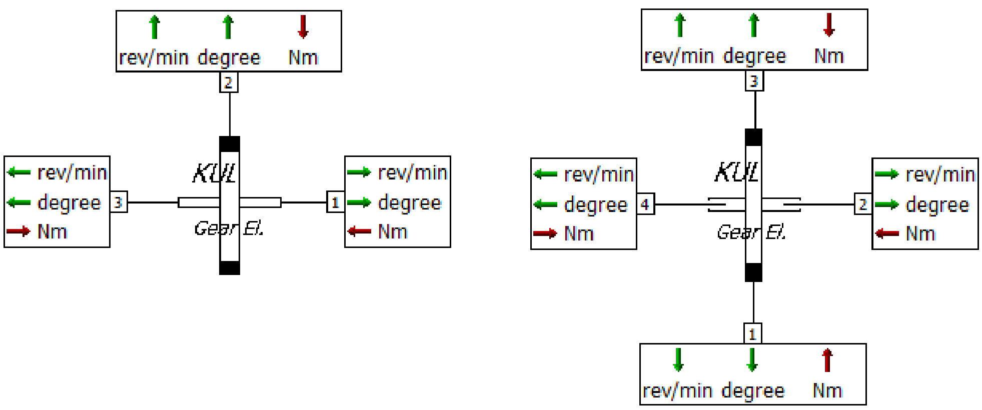


Figure 2: gear contact pair driven by a constant torque source
and connected to a damper fixed to the ground on the output

To ensure optimal model reuse and modularity, all Amesim models are build according to a dedicated auto-generated C-code skeleton in the Amesim scripting environment (Ameset). This is achieved after defining the I/O-ports, the states and all other required variables that need to be accessed from the simulation environment. This skeleton requires the user to fill a set of predefined functionalities. The first is the initialization routine that is used to start-up the simulation (e.g. to load the LUT from the specified files). Here all the preprocessing is done and each gear contact submodel instance will be initialized independently according to an object-based storage approach (i.e. beside the datasets required from every instance of the current module, a global storage identifier is created as well as a set of auxiliary structure used at runtime to discriminate which is the current instance to be executed), which is crucial in order to execute different instances of the same submodules. The most important routine to be defined implement the calculation



(a) (b)

Figure 3: Two variants (i.e. one (a) or two (b) contact interfaces) of the proposed gear inertial module

function, which is called at run-time to evaluate the module output. Here, the dedicated unique identifier is used to discriminate among the different instances of the submodule, such that the specific LUT is used to evaluate the model outputs. The last function to be filled by the user regards the termination routines, which are demanded to properly deallocate all the memory buffers previously initialized. By following the above described methodology two different modules have been created: the gear inertial submodule (Figure 3), and the gear contact module (Figure 4).

Figure 3 illustrates the elements of the gear inertial module for both its two variants: 3 and 4 ports. These gear inertial elements were developed as minor extension of the original description which is already available in the library. The main difference regards the port which connects to the contact element which in this case transmit rotational variables. The other ports are meant to be connected to spring elements that represent the flexibility of the shaft. The main difference between the 3-port (Figure 4-a) and 4-port (Figure 4-b) variants is that the additional port allows the gear to act as an intermediate gear which transfer power over two stages, although a no coupling assumption was made between the two interconnected gears.

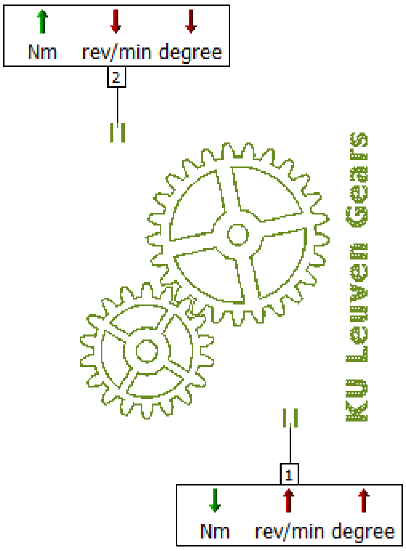


Figure 4: Gear contact module implemented in Amesim

Figure 4 shows the gear contact element whose interface allows the transmission of rotational variables. It is worth noting that the standard gear module available in Amesim only includes the linear penetration and force variables of the contact. The main role of this newly developed module is to compute the operational Torque transmitted between the gears, as a function of the angular positions of the two interconnected gears by means of an interpolation scheme over the LUT database selected for the desired gears pairs working conditions (the LUT is assigned as internal parameter of the block).

At run-time, the following steps are executed in order to compute the operational Torque:

- The relative position within the current angular pitch is deduced according to the following:

$$\alpha = mod\ (\theta_1, \tau)\ \Rightarrow\ s = \frac{\alpha}{\Delta\tau} \Rightarrow \begin{cases} i_p = \lfloor s \rfloor \\ w_p = 1 - (s - i_p) \end{cases} \qquad (6)$$
$$where: \Delta\tau = \frac{\tau}{N_p}$$

  Where the angular position within the current angular pitch, α, is obtained and rescaled in term of the current fraction, s, of angular pitch steps (being $\Delta\tau$ the width of each angular pitch step). From s, the integer part is taken as the index i_p, while the not integer part is used to retrieve the weight w_p, to be used later for interpolation purposes.

- Starting from the 2D LUT dataset defined in (5), an auxiliary 1D LUT is defined whose value are obtained as a linear combination of two columns of the original dataset:

$$\Delta\theta_{aux}(i_t) = w_p \cdot \Delta\theta_{aux}(i_t, i_p) + (1 - w_p) \cdot \Delta\theta_{aux}(i_t, i_p + 1) \tag{7}$$

- Finally, a linear interpolation is realized on the auxiliary LUT, which defines a piecewise linear function relating the Torque and the STE. Evaluating (5) using the current values of the angular positions of the two gear, the current STE is evaluated. Thereafter, the corresponding Torque is retrieved by means of linear interpolation over the auxiliary LUT.

3 Results

This section describes the numerical results produced with the presented approach in order to discuss required computational time and solution accuracy. In particular we illustrate how the method is capable to capture the main behavior of the system (mainly through the analysis of the transmission error) for varying operating and manufacturing conditions. The transmission error, already defined in section (2.1.2), is indeed extremely relevant in order to analyze the noise and vibration behavior of a geared transmission.

All the results were produced on a laptop equipped with a Intel i7-4610M CPU, 3.0 GHz and 16 GB RAM. During the preprocessing phase Matlab 2014 was used extensively. Moreover, SIEMENS LMS Imagine.Lab Amesim and Ameset were used to implement a proof-of-concept of the methodology presented along the paper.

Among the different test cases we used the following set of gears

Quantity [Unit]	G1	G2	G1m
Number of teeth [-]	**21**	**31**	**21**
Normal modulus [mm]	**4**	**4**	**4**
Normal pressure angle [deg]	**20**	**20**	**20**
Profile shift coefficient [-]	**0**	**0**	**0**
Face width [mm]	**20**	**20**	**20**
Normal backlash	**0**	**0**	**0**
Addendum modification factor* [-]	**1**	**1**	**1**
Dedendum modification factor* [-]	**1.25**	**1.25**	**1.25**
Protuberance amount [mm]	**0**	**0**	**0**
Protuberance angle [mm]	**0**	**0**	**0**
Radius of rounded corner* [mm]	**0**	**0**	**0**
Tip relief at the top circle (dNa) [mm]	**-**	**-**	**0.005**
Length of the tip relief [mm]	**-**	**-**	**See table 2**
Grinding allowance	**-**	**-**	**0**
	0	**0**	**0**
	0	**0**	**0**
Inner diameter of the gear [mm]	**60**	**60**	**60**

*of the basic rack

Table 1: Gears geometry

Damper rating was set to 25 Nm/rpm, for all the considered test cases.

3.1 Case 1: one spur gear pair

The transmission simulated in this example is represented in figure 1. It consists in a dynamic simulation of a pair of identical gears, whose manufacturing parameters are presented in table 1 – G1. Moreover the driving gear is subjected to an external constant torque, while the driven gear is connected to a damper fixed to the ground and characterized by a damper rating of 25 Nm/rpm.

The time required for the pre-processing part in Matlab, therefore for assembling the lookup table, was about 7 minutes. The angular pitch has been divided in 80 parts (so 81 columns in the lookup table) and 4 level of torques have been simulated, respectively 0, 125, 250 and 500 Nm (so 4 rows in the lookup table). For the processing part in Amesim, a print interval of 0.01 sec and a simulation time of 10 sec have been chosen, while the input torque on the driving gear has been set at 150 Nm, 300 Nm and 450 Nm. Regarding the computational time of the processing stage for a dynamic simulation in Amesim, it varied from 0.19 s when the input torque was 150 Nm to 0.39 sec when the input torque was 300 Nm.

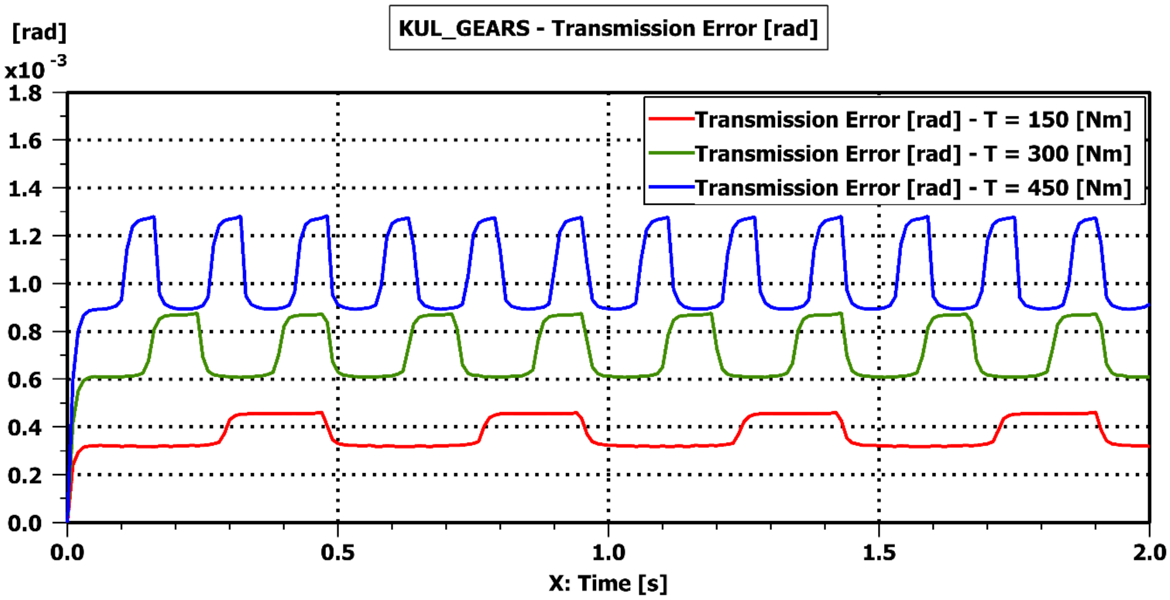


Figure 5: TE curves by varying the input torque level

Figure 5 shows the different transmission error (TE) curves calculated for this case study by setting the input torque at 150 Nm, 300 Nm and 450 Nm. The graph clearly shows how the proposed method is able to capture the classical trapezoidal shape of the TE: such shape is due to the varying stiffness of the transmission when a different number of teeth pairs come into contact at the same time. In this case for each curve, the higher value of the TE correspond to a single teeth pair in contact while the lower value to two pairs in contact. The higher frequency of switching between one and two pairs in contact that occurs at a higher torque level is consistent with the system characteristic, since for a higher input torque the system stabilizes at a higher rotational velocity when the steady state condition is reached. Moreover it is important to observe how the method is able to capture the increase in amplitude of the TE accordingly to the increase of the applied load. Such amplitude, often referred as peak-to-peak value is about 0.4 rad for the highest level of simulated torque (450 Nm) while it is about 0.2 rad for the curve corresponding to 150 Nm. Finally it is relevant to notice that also the mean value of the TE increases when the torque level increases. The mean value of the TE is indeed influenced by the gear body characteristic and it is indeed consistent with the fact that a linear FE model has been used for the gear representation that such mean value increases with the applied load.

3.2 Case 2: System level interactions over a simple transmission

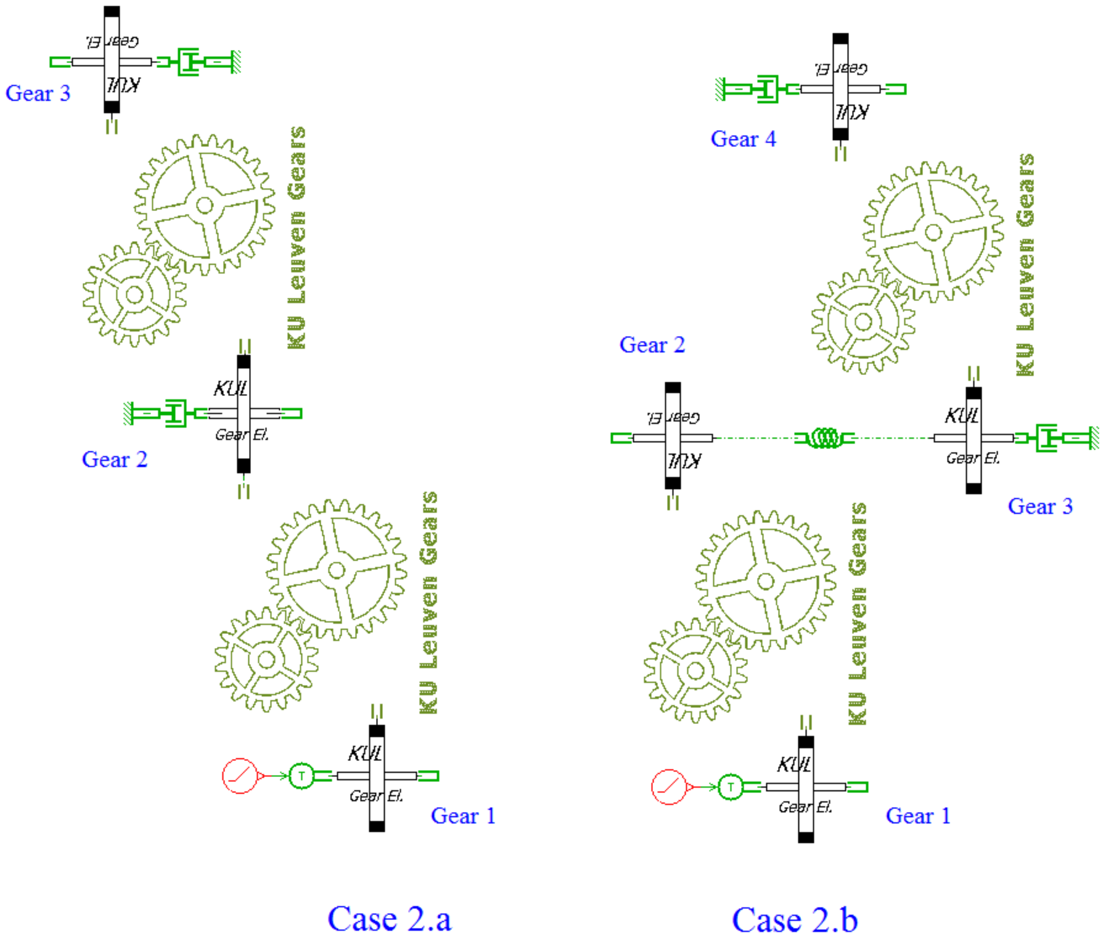


Figure 6: System level interactions over a simple transmission

In the second example the two systems represented in figure 6 are analyzed. The transmission of the left-hand side of figure 6 (case 2.a) is composed by three gears, whose characteristics are illustrated in table 1 and are respectively G1 for the first gear, G1 for the second gear and G2 for the third gear. Moreover the second gear is in this case making contact on two sides and it is therefore represented by a 4 ports gear element (illustrated in section 2). The transmission on the right-hand side of figure 6 (case 2.b) is composed by 4 gears whose characteristics are presented in table 1 and are G1 for the first three gears and G2 for the forth gear. Furthermore each gear is in this case represented by a 3 ports element but gear 2 and 3 are connected by a linear rotational spring that models the shaft on which they are mounted.

All the dampers present in case 2.a and 2.b have the same damper rating of 25 Nm/rpm and they are fixed to the ground.

Figure 7 shows the TE curves calculated for case 2.a and for case 2.b by varying the stiffness characteristic of the rotational spring between gear 2 and gear 3 of case 2.b. In particular the spring stiffness values set for the analysis were 1e5 Nm/rad, 1e6 Nm/rad and 1e8 Nm/rad. The input torque is maintained constant for all the simulation and has a value of 250 Nm.

The TE curves represent in this case the behavior of the whole geared transmission and therefore they are influenced by both gear pairs and by the characteristics of the shaft if present. In particular we can observe in figure 7 that, when the shaft stiffness is particularly high (1e8 Nm/rad), its contribution to the TE of the system becomes negligible and the result matches with the one of case 2.a. By decreasing the shaft stiffness to value of the same order of magnitude of the mean value of the gear pair stiffness or even lower, we observe (blue and green curves of figure 7) how the system TE rapidly increases. Consistently with the model the shape of the TE and its peak-to-peak value are not influenced by the shaft stiffness, since they are only function of the geometrical characteristics of the gear pairs in contact.

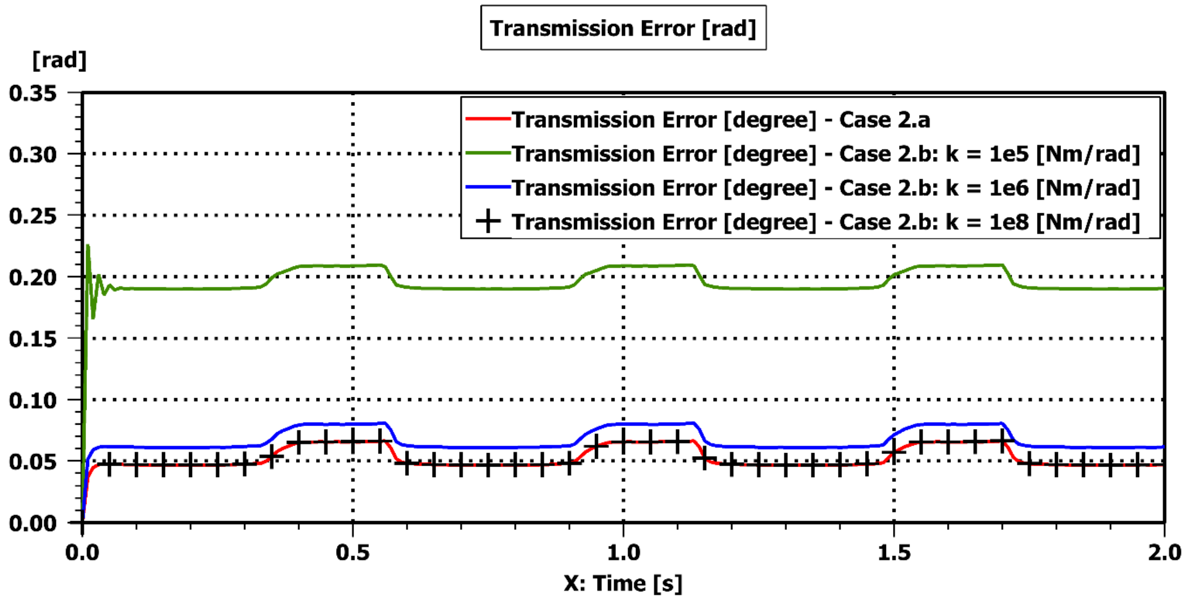


Figure 7: TE curves for a simple transmission with and without flexible shaft effects

3.3 Case 3: Gear geometry modifications effects

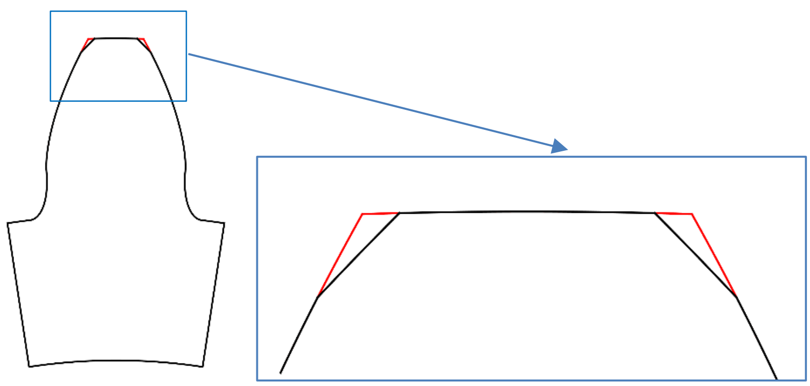


Figure 8: example of a tip-relief profile modification

In this case study the effects of gear profile geometry modification is analyzed. In particular we want to show how the method is capable of capturing the different behavior of a geared transmission when profile modifications are adopted rather than discussing the why and how such modifications are manufactured. An example of the applied profile modification can be seen in figure 8 that illustrated the different profile of the modified gear tooth with respect to the standard profile employed in the previous cases.

The system adopted in this case study is illustrated in figure 1 but both gears have been modified accordingly to table 1 (G1m) and table 2.

	a	b	c
Length of the tip relief [mm]	1.0	1.5	2.0

Table 2: Different values for tip relief geometry analysis

Figures 9 and 10 show the TE curves for two levels of torque (80 Nm and 130 Nm) calculated by varying the length of the tip relief at the top circle.

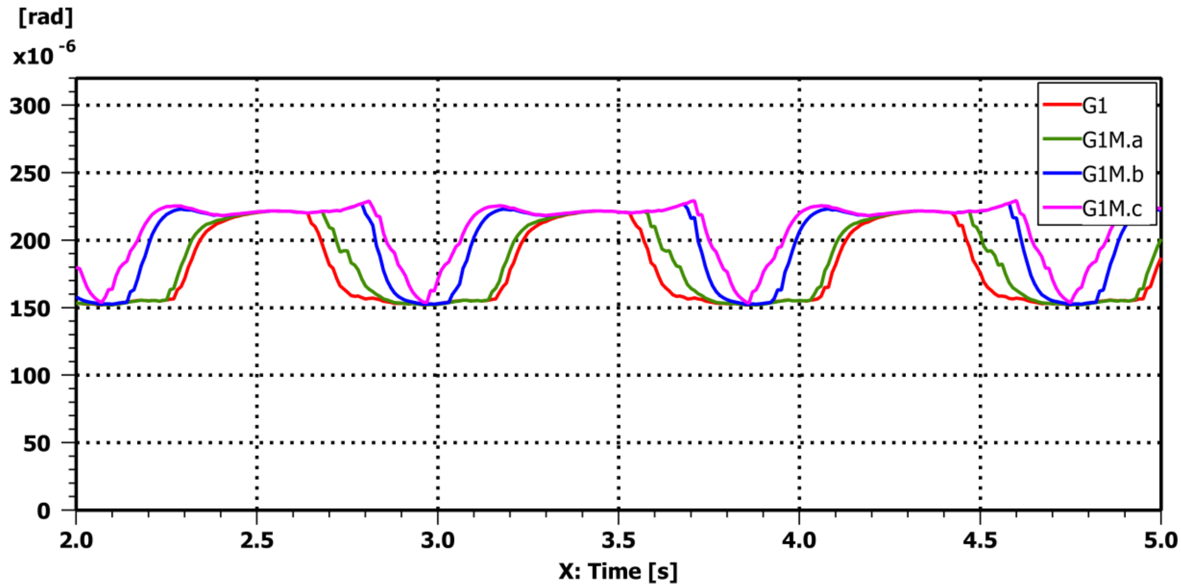


Figure 9: Tip relief analysis for the different gears:
transmission error [rad] for a low input torque of 80 [Nm]

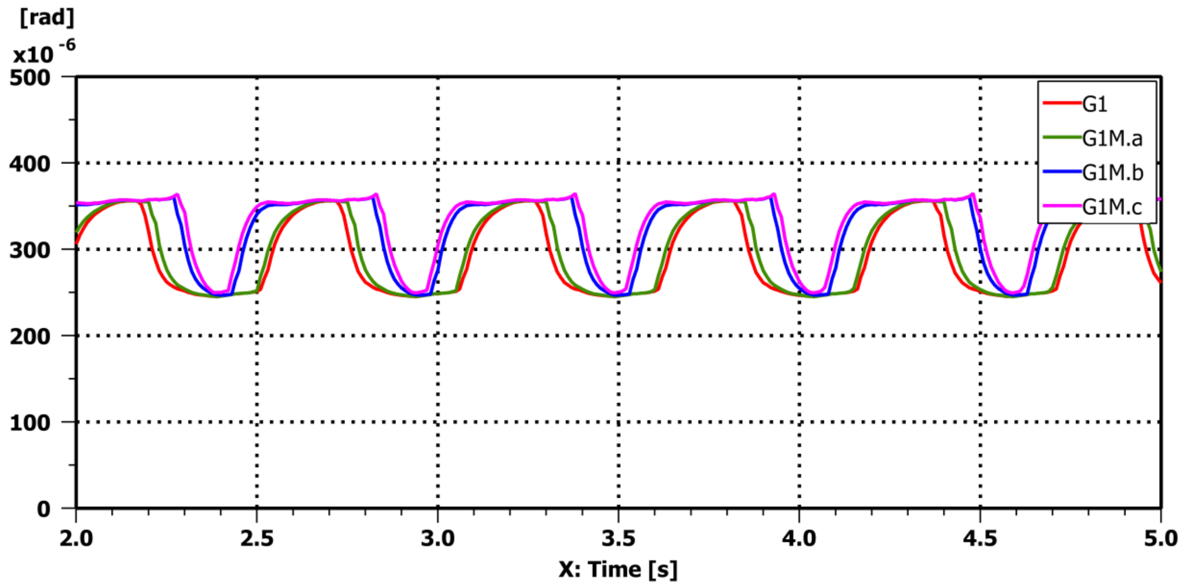


Figure 10: Tip relief analysis for the different gears:
transmission error [rad] for an input torque of 130 [Nm]

The two figures clearly illustrate the capability of the method to capture the effects induced by profile modifications. Moreover the difference with respect to the unmodified gear curves (G1) highlights the importance on taking such effects into consideration in order to properly model the system behavior of a geared transmission. Indeed from the different curves for both torques we can notice how the shape of the TE changes, therefore the different length for the period of one or two pairs of teeth in contact at the same time. Finally we can also observe that not only the peak-to-peak value of the TE changes but also its mean value.

3.4 Case 4: Gear body influences

The last case study aims to show that the proposed method is able to efficiently capture the influence of the gear body characteristics on the system behavior. In particular we will analyze the system of figure 1 of a pair of identical gears whose characteristic can be found in table 1 – G1. In this case the driving gear is modified and the width of its rim (identified with “x” in figure 10) decreases from 20 mm to 18 and 17 mm. Such thinner part extends smoothly up to 30 mm in radial direction from the gear center to become again 20 mm width like gear teeth width.

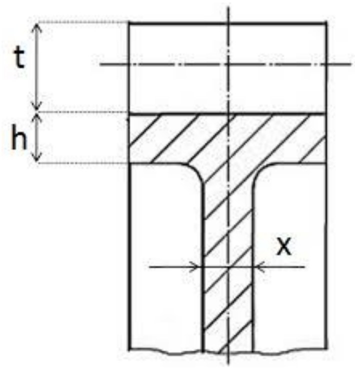


Figure 11: scheme of gear body parameters

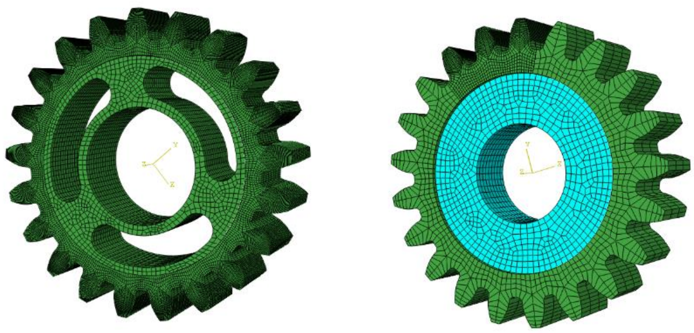


Figure 12: examples of different gear body geometries

For this example the input torque is maintained constant at a value of 250 Nm while the damper rating is held fixed of 25 Nm/rpm. Figure 13 presents the TE curves varying the width “x” of the gear body. From the graph it can be noticed the increase of the TE value accordingly to the decrease of the body width. Such behavior is easily understandable and can be justified as the reduction of the gear body torsional stiffness owing to the decreased width. Moreover the peak-to-peak value is not influenced by the body dimensions and it remains constant in all curves.

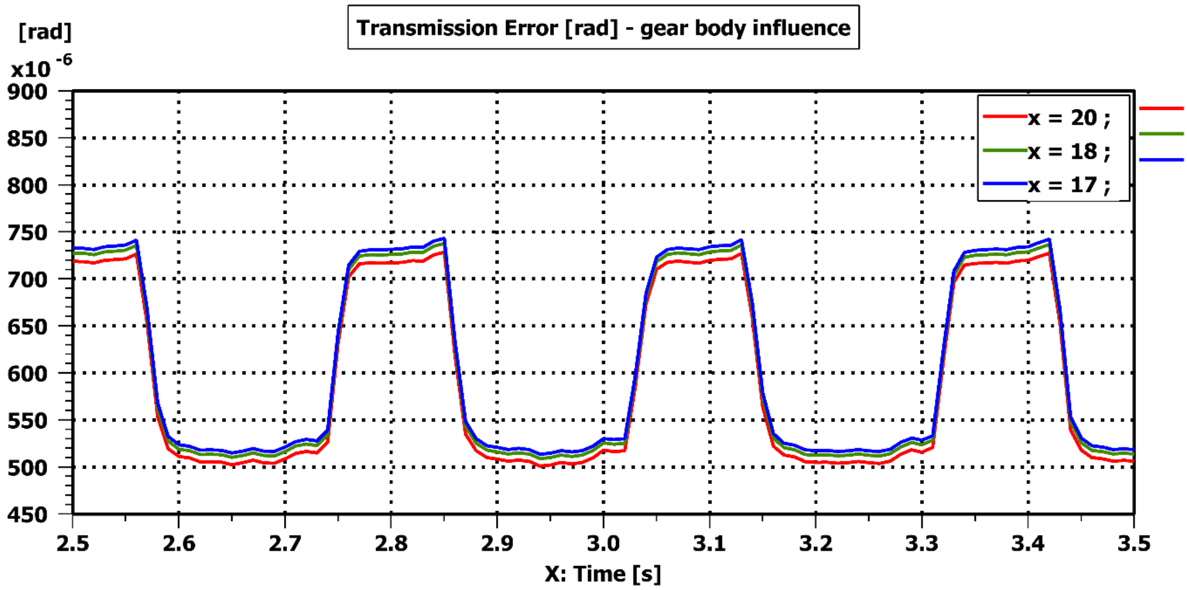


Figure 13: TE curves for different gear body widths

Therefore the method is able to capture also the gear body characteristics and their effect on the TE and it can be applied to any arbitrary gear body geometry, such as the lightweight gear with holes shown at the left-hand side of figure 12.

4 Conclusions

Recent advances in the field of model order reduction schemes posed the base for a new frontier of dynamic 3D simulation capabilities where different operational conditions can be analyzed effectively for both gears [4] and bearing components [5].

This paper aimed at integrating some of these techniques into a commercial multi-domain 1D simulation software package, defining a successful methodology able to embed precomputed detailed information originating from the high-fidelity nonlinear Finite Element simulations into a multi-domain environment for system level analysis of mechatronic drivelines. As an alternative, the methodology proposed along the paper could also be tuned in order to use experimental datasets as the base for the LUT dictionary used at runtime.

The presented methodology was tested with four different scenario, always indicating the ability to bring into the 1D modelling environment the typical 3D effects like micro/macro geometry modification, misaligned mounting, and perturbed loading conditions. However, current methodology is yet not ready to be applied for multi-optimization problems, for which the curse of dimensionality related to the expansion of the LUT on more than 2 dimensions will immediately impact feasibility.

Further research efforts are required in order to relax the quasi-static assumption hypothesis, thus leading to a more flexible interaction between the 1D modelling framework, and the model order reduction scheme used in the background to still capture the required 3D information.

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